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(54) **A condenser.**

(57) A condenser particularly for use in automobile air conditioning system, the condenser including a pair of headers (3,4) having their inner spaces divided by partitions (10,11) so as to form a cooling medium flow path in a zigzag pattern including an inlet side group of paths and an outlet side group of paths, side group of paths is 30 to 60% that of the inlet side group of paths.

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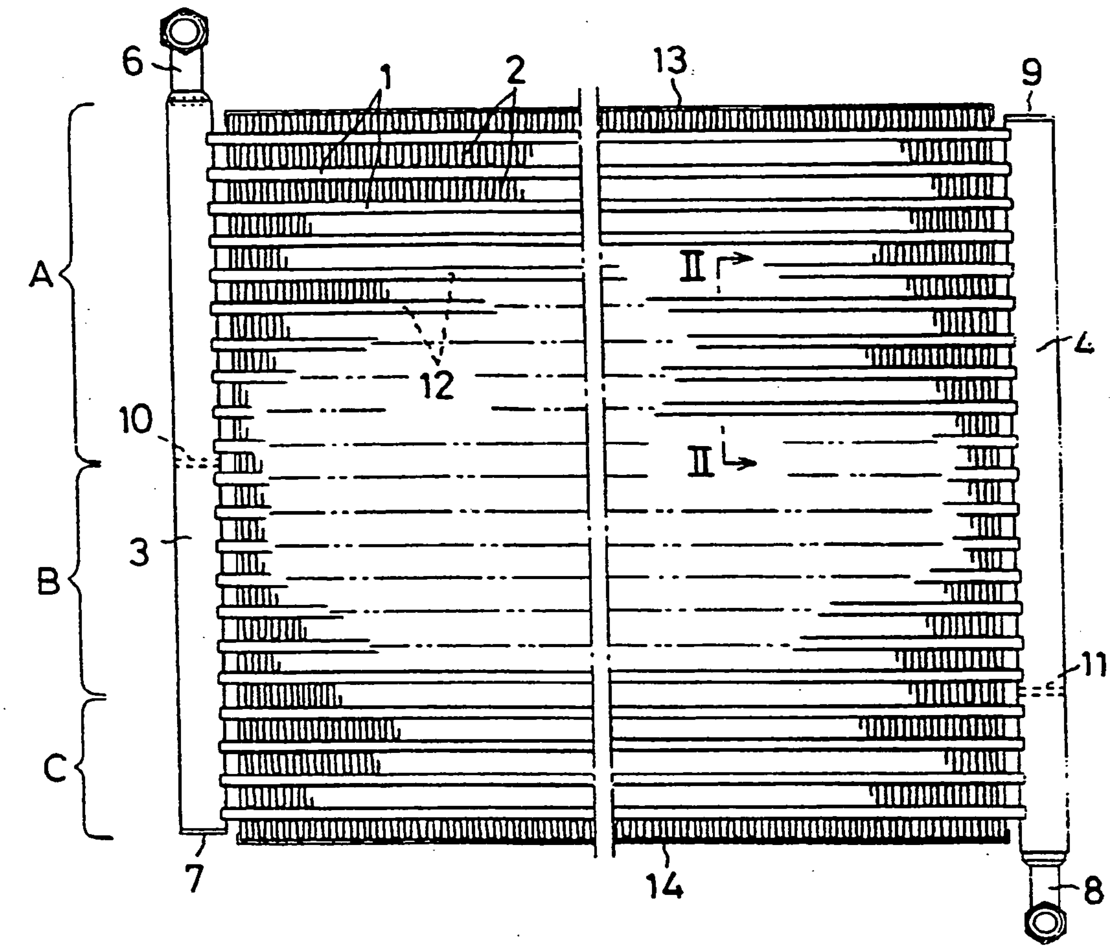

FIG. 1

BACKGROUND OF THE INVENTION

The present invention relates to a condenser particularly adapt d for use in automobile air conditioning systems.

For such use, a "serpentine" type of condenser is well known and widely used, in which is made up of a multi-bored flat tube, commonly called "harmonica" tube, bent in zigzag form, and corrugated fins sandwiched between the bent tube walls. In this way a core is constituted.

The cooling medium path in a condenser is roughly classified into two sections, that is, and inlet side section and an outlet side section. In the inlet side section the cooling medium is still in a gaseous state, and in the outlet side section it becomes liquid. In order to increase the eficiency of heat exchange the area for heat exchange of the inlet side paths should be as large as possible. On the other hand, that of the outlet side paths can be relatively small.

Since the "serpentine" type condenser consists of a single cooling medium path provided by a single pipe, an increase in the area for heat exchange in the inlet side section increases that of the outlet side section. As a whole the size of the condenser become large.

The inventors have made an invention relating to a "multi-flow" type condenser instead of the serpentine type, which is disclosed in Japanese Patent Publication (unexamined) No. 63-34466. The multi-flow type condenser includes a plurality of tubes arranged in parallel and corrugated fins sandwiched therebetween, and headers connected to opposite ends of the tubes. The headers have partitions which divide their inner spaces into at least two sections including an inlet side group of paths and an outlet side group of paths, thereby causing the cooling medium to flow in at least one zigzag pattern. The total cross-sectional area of the inlet side group of paths progressively diminishes toward the outlet side group. In this way the inlet side section has an optimum area for accommodating the cooling medium in a gaseous state, and the outlet side section has an optimum area for accommodating that in a liquid state. Thus the multi-flow type condenser has succeeded in reducing the size of condensers without trading off the efficiency of heat exchange. However, one problem arises in what proportion the whole path is divided into the gaseous phase side (i.e. the inlet side section) and the liquid phase side (i.e. the outlet side section) by partitions. The improper proportion unfavorably affects the efficiency of heat exchange and causes pressure loss on the flow of the cooling medium.

If the area in the outlet side section is insufficiently reduced as compared with that of the inlet side section, it becomes difficult to secure a sufficiently increased cross-sectional area of the inlet side section. As a result the cooling medium undergoes a larger pressure loss, and the efficiency of heat exchange decreases because of the relatively small area for heat exchange. If, however, the area in the outlet side section is excessively reduced as compared with that of the inlet side section, pressure loss is likely to increase on the flow of the cooling medium. The area for heat exchange of the inlet side section becomes too large, thereby slowing down the flow rate of the cooling medium.

Accordingly, it is an object of the present invention is to provide a condenser having cooling medium paths divided in an inlet side section and an outlet side section in an optimum proportion, thereby increasing the efficiency of heat exchange and reducing the pressure loss of a cooling medium.

Other objects and advantages of the present invention will become more apparent from the following detailed description, when taken in conjunction with the accompanying drawings which show, for the purpose of illustration only, one embodiment in accordance with the present invention.

According to the present invention, a condenser comprises a core and a pair of headers disposed parallel with each other, the core comprising: a plurality of tubes connected at ends thereof to the headers in fluid connection therewith and corrugated fins each disposed in an air flow path formed between the tubes wherein the headers are cylindrical pipes, is characterized in that the tubes are flat tubes each having inside thereof at least one reinforcing wall extending longitudinally of the flat tubes, the flat tubes each having their ends inserted in slits of the headers and soldered thereto in liquid-tight state, that each flat tube has the following dimensions:

width : 6.0 to 20 mm
height : 1.5 to 7.0 mm
height of each cooling medium path : 1.0 mm or more;

that each corrugated fin has the following dimensions:

height : 6.0 to 16 mm
fin pitch : 1.6 to 4.0 mm;

and that the inn r spaces of the headers are divided respectively by partitions to form the cooling medium paths such that the cooling medium flows zigzag within the core.

The invention will now be described further, by way of example, with reference to the accompanying drawings, in which:-

Fig. 1 is a plan view of a condenser according to the present invention:

Fig. 2 is a cross-sectional view on an enlarged scale taken along th line II II of Fig. 1;
Fig. 3 is an exploded perpsectiv view of the condenser of Fig. 1;
Fig. 4 is a fragmentary cross-sectional view on an enlarged scale showing the flat tube and the corrugated fin when observed in the same direction as in Fig. 3;
Fig. 5 is a fragmentary front view showing a relationship between the corrugated fins and the flat tubes;
Fig. 6 is a diagrammatic view showing flow patterns of a coolant medium;
Fig. 7 is a graph showing a relationship between the ratios of cross-sectional area of the outlet side section to the inlet side section and the rate of heat exchange;
Fig. 8 is a graph showing a relationship between the ratios of cross-sectional area of the outlet side section to the inlet side section and the pressure loss on the cooling medium;
Fig. 9 is a graph showing a relationship between the number of cooling medium paths and the rate of heat exchange;
Fig. 10 is a graph showing a relationship between the number of cooling medium paths and the pressure loss on the cooling medium;
Fig. 11 is a graph showing a relationship between the number of cooling medium paths, the rate of heat exchange and the pressure loss on the cooling medium;
Fig. 12 is a graph showing a relationship between the widths of flat tubes and the rate of heat transfer;
Fig. 13 is a graph showing a relationship between the heights of flat tubes and the pneumatic pressure loss;
Fig. 14 is a graph showing relationships between the rate of heat exchange and the heights of corrugated fins, and between the pneumatic pressure loss and the heights of corrugated fins; and
Fig. 15 is a graph showing relationships between the rate of heat exchange and the pitches of corrugated fins, and between the pneumatic pressure loss and the pitches of corrugated fins.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENT

Referring to Figs. 1 to 6, the illustrated condenser includes a plurality of flat tubes 1 stacked in parallel and corrugated fins 2 sandwiched between the flat tubes 1. The terminating ends of the flat tubes 1 are connected to headers 3 and 4.

Each flat tube is made of extruded aluminum, having a flat configuration as clearly shown in Figs. 2 to 4. Alternatively, the flat tubes can be multi-bored flat tubes, commonly called "harmonica tube" or else, electrically seamed tubes can be used.

Each corrugated fin 2 has a width identical with that of the flat tube 1. The fins 2 and the flat tubes 1 are brazed to each other. Preferably the fins 2 are provided with louvers 2a on the surface.

The headers 3, 4 are made up of electrically seamed pipes of aluminum, and each have holes 5 of the same shape as the cross-section of the flat tubes 1 so as to accept the tube ends 1a. The inserted tube ends 1a are brazed in the holes 5. As shown in Fig. 1, the headers 3 and 4 are connected to an inlet pipe 6 and an outlet pipe 7, respectively. The inlet pipe 6 allows a cooling medium to enter the header 3, and the outlet pipe 8 allows the used cooling medium to discharge. The headers 3 and 4 are closed with covers 7 and 9, respectively. The reference numerals 13 and 14 denote side places attached to the outermost corrugated fins 2.

The header 3 has its inner space divided by a partition 10 into two sections, and the header 4 also has two sections divided by a partition 11. In this way the whole cooling medium path 12 is divided into an inlet side group (A), an intermediate group (B) and an outlet side group (C) as shown Figs. 1 and 6. The cooling medium flows in zigzag patterns throughout the groups (A), (B) and (C). As shown in Fig. 6, it is arranged that the intermediate group (B) has a smaller number of flat tubes 1 (that is, paths) than the inlet side group (A), which means that the cross-sectional area of the intermediate group (C) of paths is smaller than that of the group (A). It is also arranged that the outlet side group (C) has a smaller number of flat tubes 1 (that is, the number of cooling medium paths) than the intermediate group (B), which means that the cross-sectional area of the outlet side group (C) of paths is smaller than that of the group (B).

In terms of percentage the entire cross-sectional area of the outlet side group (C) is 30 to 60% of that of the inlet side group (A). If the percentage is less than 30%, the cross-sectional area of the outlet side group (C) becomes small to increase the pressure loss in the cooling medium. At the same time, the cross-sectional area of the inlet side group becomes large to slow down the flow rate of the cooling medium, thereby reducing the efficiency of heat exchange. If the percentage exceeds 60%, the cross-sectional area of the inlet side group (A) becomes small to increase the pressure loss in the cooling medium. In addition, the area for heat transfer is reduced, th reby reducing the efficiency of heat exchange. It is more preferred that the entire cross-sectional area of the outlet side group (C) is 35 to 50% of that of the inlet side group (A). As shown in Figs. 7 and 8, this more restricted range exhibits the highest efficiency of heat exchange and the lowest pressure loss in the cool-

ing medium.

As shown in Fig. 6, the cooling medium is introduced into the inlet side group (A) through the inlet pipe 6 and flows therethrough. Then the cooling medium turns from the right-hand header 4 and enters the intermediate group (B). Then it turns from the left-hand header 3 and nters the outl t side group (C). Finally the cooling medium is discharged through the outlet pipe 8. In this way the cooling medium flows in zigzag patterns. Air enters the air paths constituted by the corrugated fins 2 in the direction (W) in Fig. 2. Heat exchange is effected between the air and the cooling medium flowing through the groups (A), (B) and (C). While the cooling medium passes through the inlet side group (A), it is still in a gaseous state and has a relatively large volume, which is effectively accommodated in the capacity provided by the paths of the group (A) and keeps contact with the flat tubes 1 in a wide range so that the gaseous cooling medium smoothly condenses and reduces its volume. When the cooling medium flows through the outlet side group by way of the intermediate group (B), it becomes completely liquid, and has such a reduced volume as to be accommodated in a relatively small cross-sectional area of the outlet side group (C). Thus the pressure loss is minimized, thereby enhancing the efficiency of heat exchange.

The illustrated embodiment has three groups (A), (B) and (C), but the number (N) of groups is not limited to it. Preferably the number (N) is 2 to 5 groups for the reason explained below:

Figs. 9 to 11 show the results obtained by experiments in which condensers having twenty-four flat tubes are employed, each having a different number of groups. A cooling medium is introduced into each of the condensers at the same flow rate. Each graph shows the resulting rate of heat exchange and pressure loss in the cooling medium, and changes in the rate of heat exchange and pressure loss with respect to the ratio of the outlet side group to the inlet side group. Throughout the experiments the inlet side group, the intermediate group and the outlet side group have the same cross-sectional area. Fig. 9 shows the rates of heat exchange achieved when the speed of wind Vf is 2m/sec and when it is 3m/sec each in front of the condenser. It will be understood from Fig. 9 that when the number (N) of the groups is less than 2 the rate of heat exchange is low, whereas when it exceeds five, the rate of heat exchange gradually diminishes. It will be understood from Fig. 10 that as the number (N) of groups increases, the pressure loss in the cooling medium increases, especially when the number (N) exceeds five, it abruptly increases. It will be understood from Fig. 11 that if the number (N) of the groups is less than two, the pressure loss is low but the rate of heat exchange is also low. Therefore the ratio of the rate of heat exchange to the pressure loss becomes low, which indicates that there is an imbalance between the pressure loss and the rate of heat exchange. If the number (N) of the groups exceeds five, the rate of heat exchange becomes relatively high the pressure loss becomes low. The ratio between them is low, thereby causing an imbalance between the pressure loss and the rate of heat exchange.

As is evident from the results of the experiments, when the number (N) of the groups is 2 to 5, the rate of heat exchange is high, and the pressure loss in the cooling medium is low. Thus the ratio between them is well balanced. As described above, it is arranged to ensure that the cross-sectional area of the outlet side group (C) is arranged to have 30 to 60% of that of the inlet side group (A). In addition, the number (N) of the group is arranged to be 2 to 5, which enhances the efficiency of the heat exchange as a result of the reduced pressure loss.

It is preferred that the width (Wt) of each flat tube 1 is in the range of 6.0 to 20mm, the height (Ht) thereof is in the range of 1.5 to 7.0mm, the height (Hp) of the cooling medium paths 12 in the flat tubes 1 is 1.0mm or more. It is also arranged that the height (Hf) of the corrugated fins 2 or a distance between the adjacent flat tubes 1 is in the range of 6 to 16mm and that the fin pitch (Fp) is in the range of 1.6 to 4.0mm. The reasons why the above-mentioned ranges are preferable will be described below:

As is evident from Fig. 12, if the width (Wt) of the flat tubes 1 is less than 6.0mm, the corrugated fins 2 sandwiched therebetween will be accordingly narrow in width. The narrow width of the corrugated fins 2 limit the size and number of the louvers 2a, which decreases the efficiency of heat exchange. If the flat tubes 1 are 20mm or more, the corrugated fins 2 sandwiched therebetween will accordingly become large. The large fins increases a drag on the flowing air. In addition, the large fins increases the weight of the condenser. It is therefore preferred that the width (Wt) of the flat tubes is in the range of 6.0 to 16mm, more preferably, 10 to 14mm.

The height (Ht) of each flat tube 1 is preferably in the range of 1.5 to 7.0mm. If it exceeds 7.0mm, the pressure loss in the air flow increases. If it is less than 1.5mm, it is difficult to increase the height (Hp) of the air paths by 1.0mm or more because of the limited thickness of the flat tubes. It is preferred that it is in the range of 1.5 to 5.0mm; mor preferably, 2.5to 4.0mm.

The height (Hp) of the cooling medium flow paths in th flat tub s 1 is preferably 1.0mm or more. If it is less than 1.0mm, the pressure loss in the cooling medium incr ases, thereby decreasing the rates of heat transfer. It is pref rred that it is in the range of 1.5 to 2.0mm.

The height (Hf) of th corrugated fins 2 is in the range of 6.0 to 16mm. If it is less than 6mm, the pressure loss in the air will increase as shown in Fig. 14. If it exceeds 16mm, th number of total fins decreas s, thereby

reducing the efficiency of heat exchange. The optimum range is 8.0 to 12mm.

As shown in Fig. 15, the fin pitches is preferably in the range of 1.6 to 4.0mm. If they are less than 1.6mm, the louvers 2a interfere with th flow of the air, thereby increasing the pressure loss in the air flow. If they exceed 4.0mm, the efficiency of heat exchange decreases. It is therefore preferred that the rang is 1.6 to 3.2mm; more preferably, 2.0 to 3.2mm.

As is evident from the foregoing description, the condensers of the present invention are constructed with the flat tubes, the corrugated fins and the headers in which the widths and heights of the flat tubes, the heights of the cooling medium flow paths, the heights and pitches of the fin are determined at optimum values, thereby reducing the pressure losses which the air and the cooling medium undergo. As a result the efficiency of heat exchanges is enhanced.

In the illustrated embodiment the cross-sectional area of the cooling medium paths 12 progressively diminishes from the inlet side group to the outlet side group through the intermediate group. However it is possible to modify it to an embodiment in which the inlet side group and the intermediate group have the same cross-sectional area which is larger than that of the outlet side group. In the illustrated embodiment the reduction in the cross-sectional area is effected by reducing the number of the flat tubes, but it is possible to reduce the cross-sectional areas of the individual flat tubes without changing the number thereof. The headers 3 and 4 are provided at their erected postures between which the flat tubes 1 are horizontally stacked one above another, but it is possible to modify it to an embodiment in which the headers 3 and 4 are positioned up and down between. which the flat tubes are vertically arranged in parallel.

Claims

1. A condenser comprising a core and a pair of headers (3, 4) disposed parallel with each other, the core comprising: a plurality of tubes (1) connected at ends thereof to the headers in fluid correction therewith; and corrugation fins (2) each disposed in an air flow path formed between the tubes (1), wherein the headers (3, 4) are cylindrical pipes, characterized in that the tubes (1) are flat tubes each having inside thereof at least one reinforcing walls extending longitudinally of the flat tubes, the flat tubes (1) each having their ends inserted in slits of the headers (3, 4) and soldered thereto in liquid-tight state; that each flat tube (1) has the following dimensions:
   width : 6.0 to 20 mm
   height : 1.5 to 7.0 mm
   height of each cooling medium path : 1.0 mm or more;
   that each corrugated fin (2) has the following dimensions:
   height : 6.0 to 16 mm
   fin pitch : 1.6 to 4.0 mm
   and that the inner spaces of the headers (3, 4) are divide respectively by partitions (10, 11) to form the cooling medium paths such that the cooling medium flows zigzag within the core.

2. A condenser comprising a core and a pair of headers (3, 4) disposed parallel with each other, the core comprising: a plurality of tubes (1) connected at ends thereof to the headers in fluid connection therewith; and corrugated fins (2) each disposed in an air flow path formed between the tubes (1), wherein the headers (3, 4) are cylindrical pipes, characterized in that the tubes (1) are flat tubes each having inside thereof at least one reinforcing walls extending longitudinally of the flat tubes, the flat tubes (1) each having their ends inserted in slits of the headers (3, 4) and soldered thereto in liquid-tight state; that each flat tube (1) has the following dimensions:
   width : 6.0 to 16 mm
   height : 1.5 to 5.0 mm
   height of each cooling medium path : 1.0 mm or more;
   that each corrugated fin (2) has the following dimensions:
   height : 8.0 to 16 mm
   fin pitch : 1.6 to 3.2 mm;
   and that the inner spaces of the headers (3, 4) are divided respectively by partitions (10, 11) to form the cooling medium paths such that the cooling medium flows zigzag within the core.

3. A condenser comprising a core and a pair of headers (3, 4) disposed parallel with each other, the core

comprising: a plurality of tubes (1) connected at ends thereof to the headers in fluid connection th rewith; and corrugated fins (2) each dispos d in an air flow path formed between the tubes (1), wherein th headers (3, 4) are cylindrical pipes, characterized in that the tubes (1) are flat tubes each having inside thereof at least one reinforcing walls extending longitudinally of the flat tubes, the flat tubes (1) each having their ends inserted in slits of the headers (3, 4) and soldered thereto in liquid-tight state; that each flat tube (1) has the following dimensions:

width	: 6.0 to 14 mm
height	: 2.5 to 4.0 mm
height of each cooling medium path	: 1.5 to 2.0 mm;

that each corrugated fin (2) has the following dimensions:

height	: 8.0 to 12 mm
fin pitch	: 2.0 to 3.2 mm;

and that the inner spaces of the headers (3, 4) are divided respectively by partitions (10, 11) to form the cooling medium paths such that the cooling medium flows zigzag within the core.

4. A condenser according to claim 1, characterized in that the corrugated fin (2) are provided with louvers (2a) on their surfaces.

5. A condenser according to claim 1, characterized in that the headers are aluminum pipes round in their cross sections.

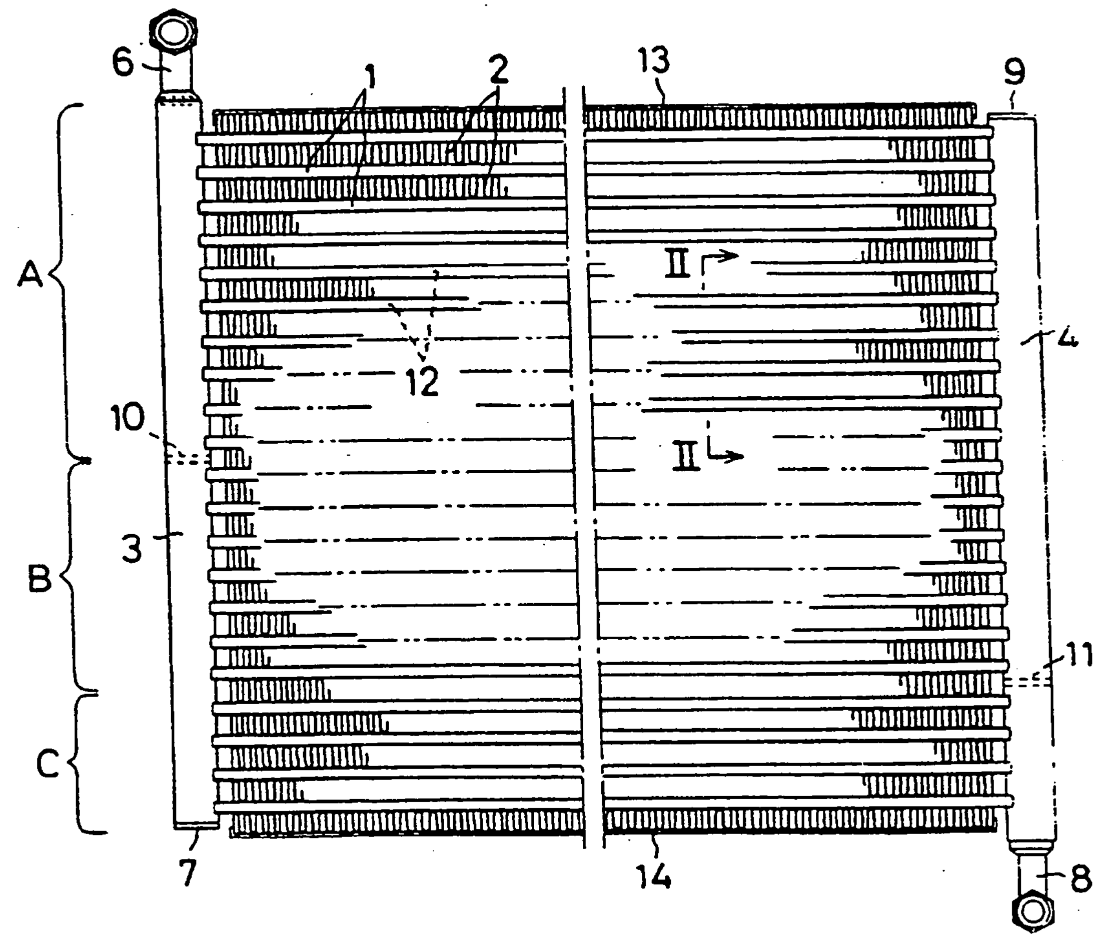

FIG. 1

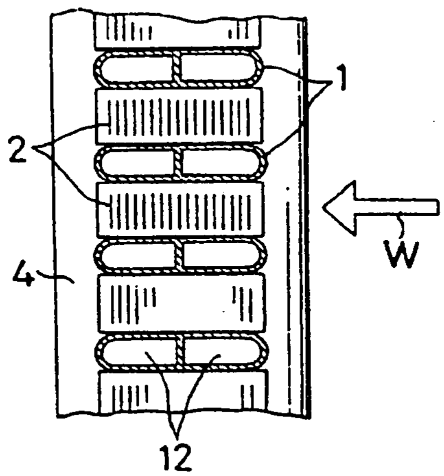

FIG. 2

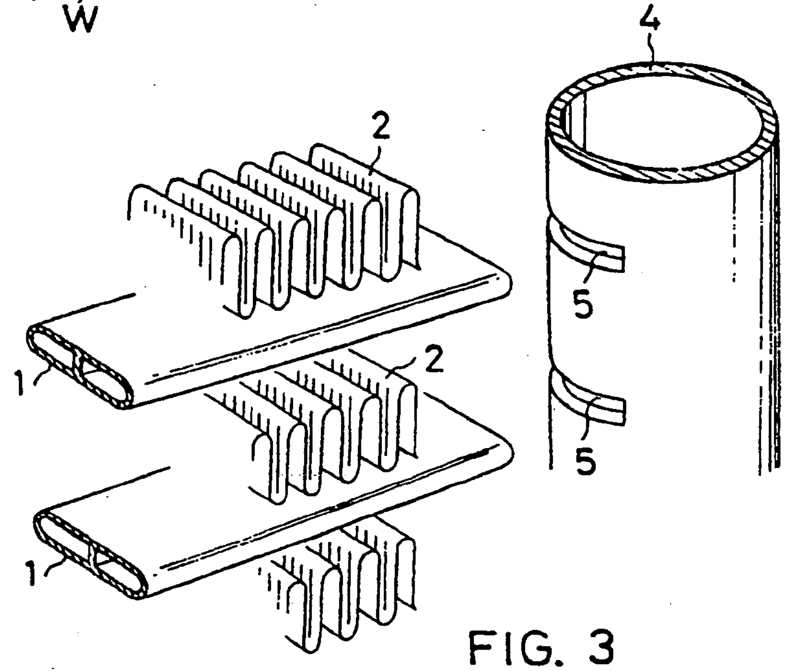

FIG. 3

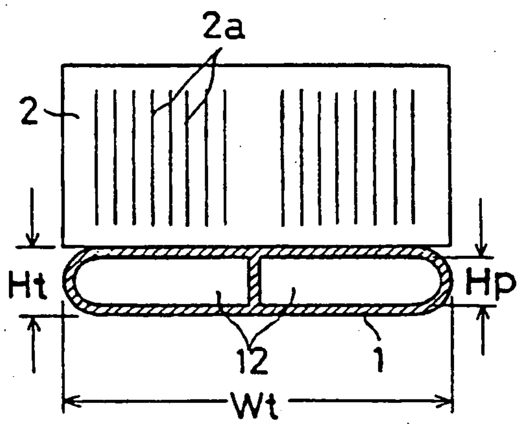

FIG. 4

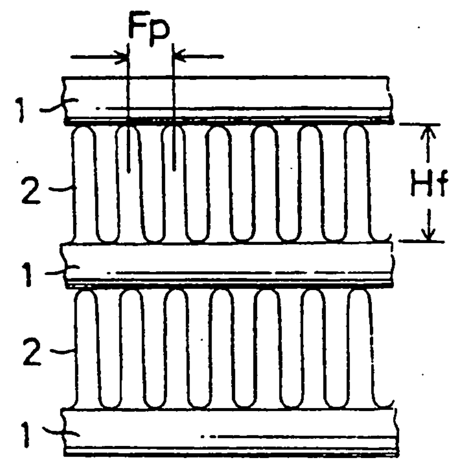

FIG. 5

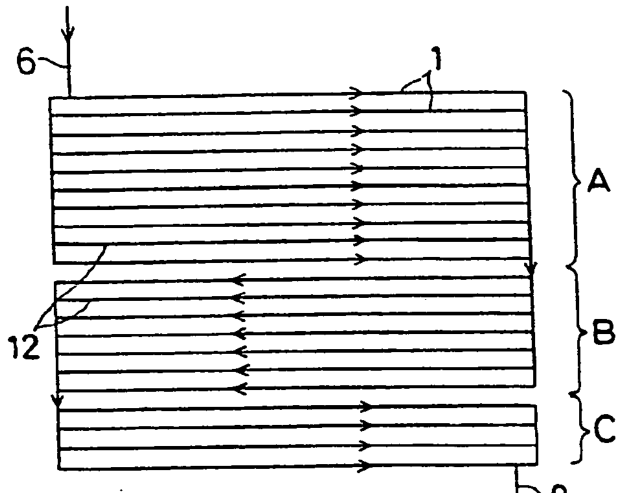

FIG. 6

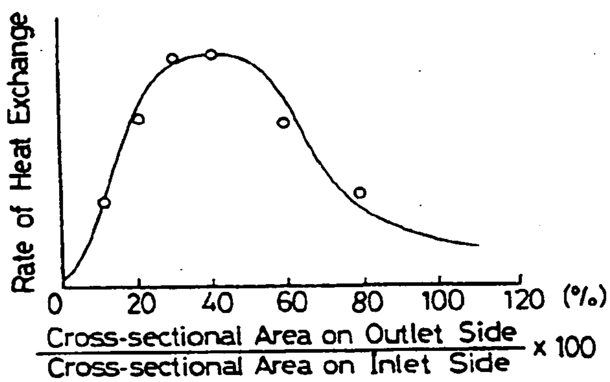

FIG. 7

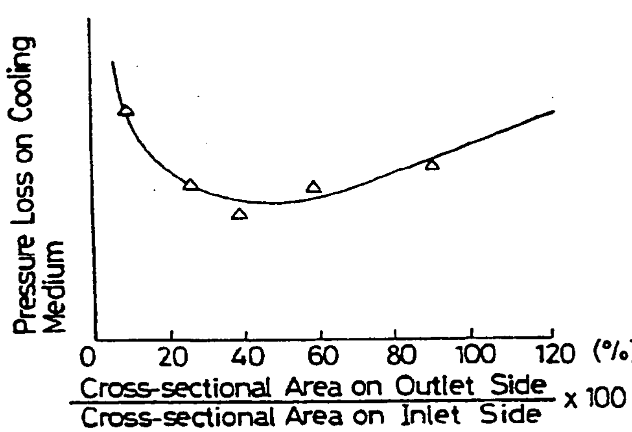

FIG. 8

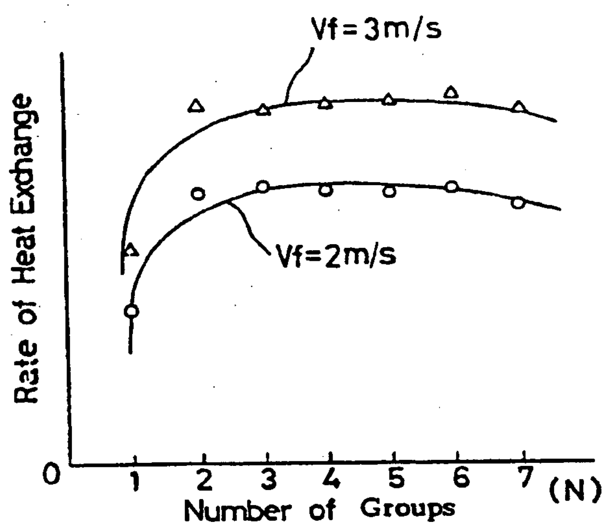

FIG. 9

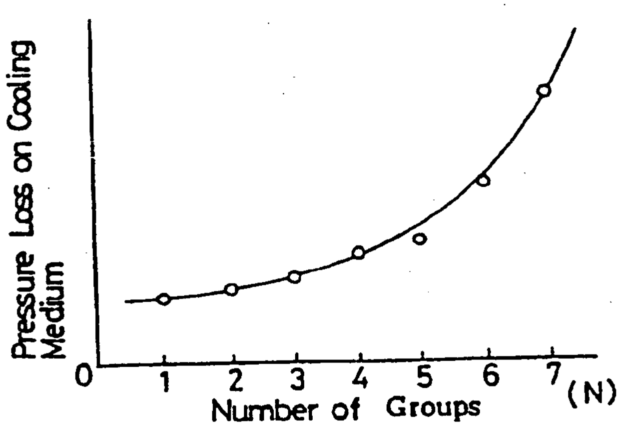

FIG. 10

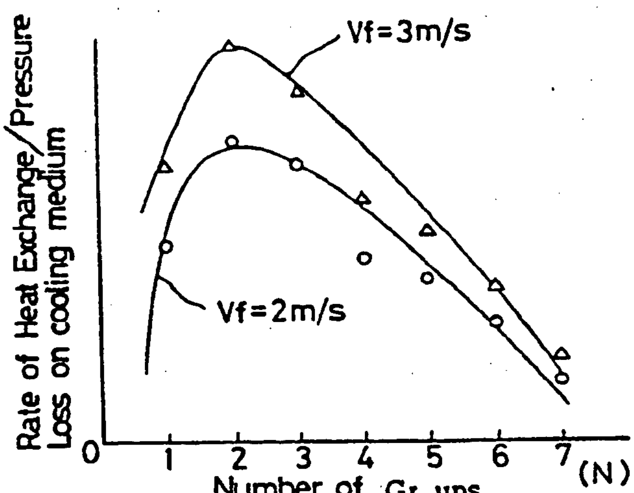

FIG. 11

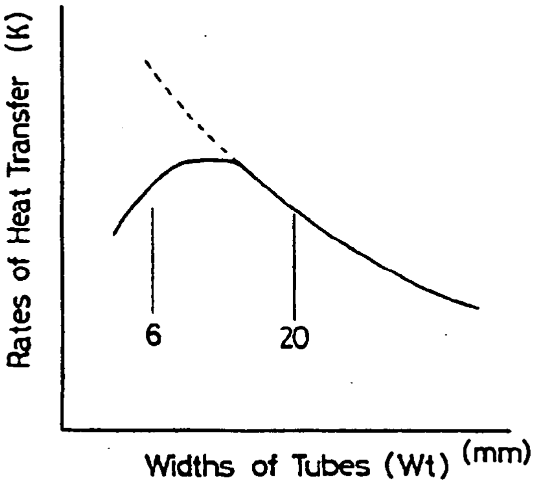

FIG. 12

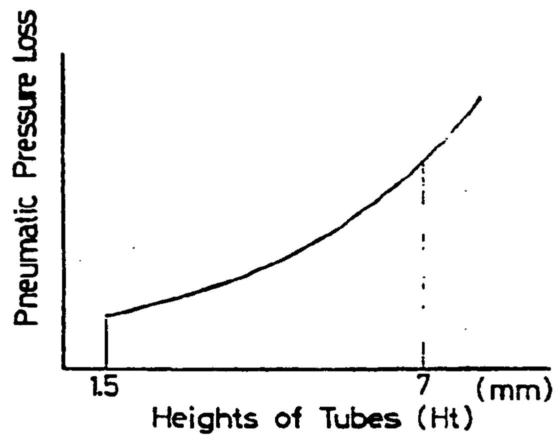

FIG. 13

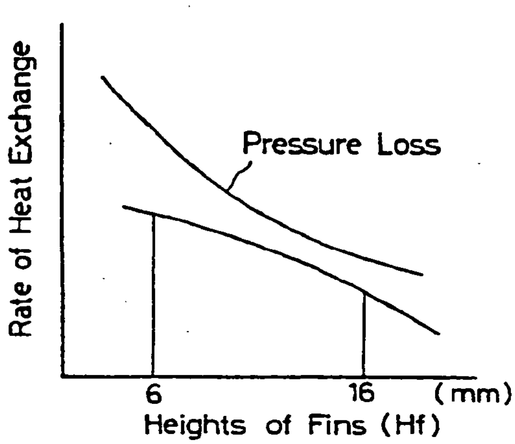

FIG. 14

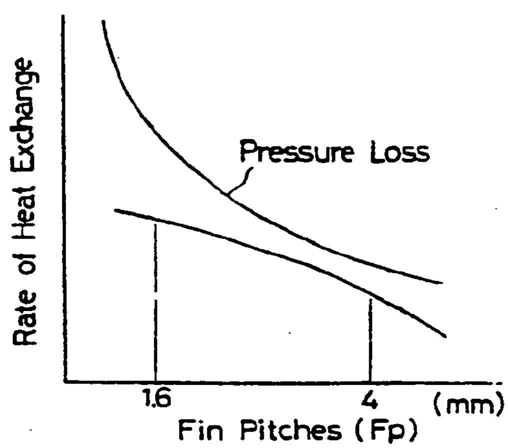

FIG. 15